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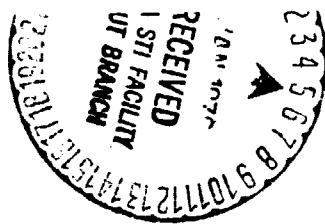
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**PRELIMINARY EVALUATION OF A HEAT PIPE
HEAT EXCHANGER ON A REGENERATIVE TURBOFAN**

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16. Abstract <p>A preliminary evaluation was made of a regenerative turbofan engine using a heat pipe heat exchanger. The heat exchanger had an effectiveness of 0.70, a pressure drop of 3 percent on each side, and used sodium for the working fluid in the stainless steel heat pipes. The engine was compared to a reference turbofan engine originally designed for service in 1979. Both engines had a bypass ratio of 4.5 and a fan pressure ratio of 2.0. The design thrust of the engines was in the 4000 N range at a cruise condition of Mach 0.98 and 11.6 km. This study showed that heat pipe heat exchangers of this type would cause a large weight and size problem for the engine. The penalties were too severe to be overcome by the small uninstalled fuel consumption advantage. This type heat exchanger should only be considered for small airflow engines in flight applications. Ground applications might prove more suitable and flexible.</p>		
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SUMMARY

E-8591

A preliminary evaluation was made of a regenerative turbofan engine using heat pipes for the heat exchanger. The heat pipes used sodium for the working fluid. The effectiveness of the heat exchanger was fixed at 0.70, and the turbine-rotor-inlet temperature and overall pressure ratio were varied from 1480 to 1810K(2660 to 3260R) and from 6 to 12 respectively. The pressure loss for the heat exchanger was assumed to be 3 percent on each side. The regenerative turbofan performance was compared to an advanced turbofan engine. Both engines had the same type two stage fan with a pressure ratio of 2.0 and the same bypass ratio. This study made no attempt to optimize the bypass ratio due to the nature of the results.

The uninstalled specific fuel consumption of the regenerative turbofan was 3.3 percent better than the reference turbofan. The heat exchanger calculations lead to the conclusion that this type of heat exchanger would not package within the radius of the low pressure turbine case exit plane. The resulting bulge in the core nacelle would force the fan nacelle to have a larger diameter resulting in a significant drag penalty. The weight of the heat exchanger was much greater than the weight saved by the lighter compressor and the other innovative component arrangements assumed in this engine. These weight penalties more than offset the better uninstalled performance and resulted in at least a 10 percent increase in the fuel used.

This type of heat exchanger should only be considered for applications where weight and size are of secondary importance. Ground applications such as power plants, ships, trains, and maybe even trucks and cars might be such applications. Any further work on heat pipe regenerators for flight applications should be limited to either small turboshaft engines where the bulge and weight are of little consequence or to engines where the core flow is very small

relative to the entire engine flow. Such an application might be a small, slow flying turboprop.

INTRODUCTION

Since 1960, considerable effort has been spent studying and developing heat exchangers for turbine engines. These efforts were aimed at determining if a regenerative turbofan or turboprop engine was feasible. The Air Force, Army, Navy, NASA, and private industry have all participated in this effort at one time or another as noted by references 1 to 5. One of the relatively successful development efforts resulted in Allison's T78-A-2 Regenerative Turboprop Engine. Probably the most complete general research effort was carried out by the Air Force at the time the C5-A military airplane was being designed, reference 5. The Air Force spent over 8 million dollars on the regenerative turbofan effort. There were five engine contractors and three airframe contractors involved at that time.

Three basic types of heat exchangers were examined in the Air Force study, gas to gas, gas to liquid, and the rotary type. One type not examined was the heat pipe heat exchanger. While studies such as references 4 and 5 were not optimistic about the use of regenerators in general and heat pipes in particular, it was hoped that recent experience with heat pipes in such applications as space satellites might shed some new light on the subject. The basic advantage of the heat pipe regenerator over other types is the rather simple way the heat exchanger can be laid out behind the core of a typical turbofan engine. Also heat pipes can be made to be very reliable, a quality not usually associated with heat exchangers.

The purpose of this report is to investigate heat pipe regenerators for a turbofan engine and compare its performance to that of a reference turbofan. In most past studies, the conclusions were weakened by the lack of actual installed engine data. The studies were usually done parametrically, and regardless of how good or bad the answers turned out, it was always indicated that an actual application would be necessary to determine the real potential. Since this lack of actual application seemed to be the stumbling block, it was decided to apply the heat pipe regenerator to an advanced but realistic study engine. The engine chosen was the Pratt & Whitney STP 429 which was a proposed 1979 advanced technology engine. The STP 429, at the time the analysis was performed for this study, (nearly two years ago) was forecast as a typical engine for future commercial transports. Its cruise Mach number of 0.98 and

other characteristics differ from more recent fuel-conservative-engine designs. However, it is felt that the conclusions of this study are still valid. The results are being published at this time in order to document an unsuccessful approach to a topic of great current interest.

The regenerative engines studied had overall pressure ratios from 6 to 12, turbine-rotor-inlet temperatures from 1480 to 1810K (2660 to 3260R) and the same fan performance and bypass ratio as the reference engine. Infact, the fan design was assumed to be unchanged from the STF 429. The only change in the low spool was in the turbine. In the regenerative engines, the number of low pressure turbine stages varied from the initial value of four on the STF 429. The regenerative engine selected for the detailed analysis had only three stages in the low turbine. The high pressure compressor was modified from the STF 429 design to optimize the cycle using the heat exchanger. The final design used a single stage radial compressor with a pressure ratio of 5 instead of the multi-stage axial one with a pressure ratio of 12.5. The heat exchangers in this study were added behind the low pressure turbine in order to transfer waste heat directly from the exhaust to the compressor exit air. This preheats the compressor exit air before the combustor and thus, less fuel is needed to raise the air to the desired turbine-rotor-inlet temperature. The heat exchanger effectiveness was fixed at 0.70 since that was a typical value as noted in reference 1 for advanced heat pipes.

SYMBOLS

A	area
Aa	fin surface area available as determined from the geometry of the problem
A _r	fin surface area required to transfer the heat
A ₁	air side frontal area of the heat exchanger
A ₂	gas side frontal area of the heat exchanger
C _p	specific heat at constant pressure
CPR	compressor pressure ratio
FFR	fan pressure ratio
h	heat transfer coefficient
k	thermal conductivity
K	degrees Kelvin
L	length of heat exchanger
m	mass flow
MN	Mach number
Nu	Nusselt number
OPR	overall pressure ratio
P	total pressure
Pr	Prandtl number
$\Delta P/P$	change in pressure/reference pressure
Q ₁	total heat actually transferred into the air
Q ₂	total heat removed from the heat pipe
R	degrees Rankine
R ₁	radius of outer shell of heat exchanger
R ₂	radius of shell that separates air and gas flow
Ra	gas constant for air
Re	Reynolds number
SFC	specific fuel consumption
t	fin spacing at radius R ₂
T	total temperature
TOGW	takeoff gross weight
v	velocity
ϵ	effectiveness of heat exchanger
s	$A_1/(A_1+A_2)$
ρ	density
μ	viscosity

Subscripts

3	conditions at air side of heat exchanger entrance
3x	conditions at air side of heat exchanger exit
4	conditions at the turbine-rotor-inlet
55	conditions at the gas side heat exchanger entrance
55x	conditions at the gas side heat exchanger exit

METHOD OF ANALYSIS

Before the heat exchanger could be designed, it was necessary to determine the proper cycle parameters. The Geneng computer program used to make the cycle calculations, reference 6, was modified to do simple design point heat exchanger problems. From previous studies such as reference 5, it was known that the optimum overall pressure ratio (OPR) was between 8 and 12. Over this range of OPR, the specific fuel consumption (SFC) was fairly constant for an effectiveness (ϵ) of 0.85. However, thrust was highly dependent on turbine-rotor-inlet temperature (T₄). Therefore, a range of T₄'s were examined from 1480 to 1810K (2660 to 3260R) along with a range of OPR's from 6 to 12. From reference 1 it was known that a typical ϵ for advanced heat pipe design was 0.70. So that was the value used in the preliminary cycle calculations. This is no reason that the ϵ could not be greater than 0.70. However, the nature of the results were such that higher levels of ϵ would have made the weight of the heat exchanger even worse while contributing better but non-offsetting improvements in SFC.

Table 1 lists the cycle and heat exchanger parameters used in the initial calculations. The baseline engine is the STP-429. Each of the other four columns in the table are distinguished by increasing T₄ at the rate of 110K (200 R) per column. At each T₄ level, compressor pressure ratio (CPR) was varied from 3 to 6. This provided the range of OPR's from 6 to 12 since the fan pressure ratio (PPR) was fixed at 2.0.

Since the turbine cooling air would now be at a lower temperature than in the reference engine, a schedule of turbine cooling bleed was used. It was based on the full film coverage method used in reference 7. The cooling for the reference engine was corrected to the same basis. The heat exchanger pressure losses were initially assumed to be 3 percent on the air side and the same on the gas side. This is consistent with reference 1 also.

Engine and Heat Exchanger Layout

From past studies it was obvious that a specific application had to be attempted in order to obtain an evaluation of the tradeoff between regenerative turbofans and nortral turbofans. Weight has also been a very big obstacle for regenerators in the past as well as simplicity

and service life. It was decided that, to have any chance of success, the simplest radial heat pipe design would have to be used. This selection was based on the results of reference 1 where C. C. Silverstein examined several different configurations and determined that the radial one was best. Counter flow was a necessity as was innovative changes within the core of the engine. Every attempt had to be made to save weight and limit the heat exchanger size.

The reference and regenerative turbofans are shown in figure 1. While these flow paths are labeled sketches, they are nearly to the same scale. The main effort on the regenerative turbofan was directed toward repackaging the core to save as much weight and space as possible. The outside lines of the engine were forced to remain essentially unchanged. Thus, the drag and interference changes should be nil. The basic core of the engine was shortened by about 1.2m (4 ft.). This was accomplished by removing the 10 stage compressor ($CPR=12.5$) and replacing it with a single stage radial compressor ($CPR=5$). The radial compressor was desired since the air flow had to be turned 90 degrees anyway and taken to the core perimeter for ducting to the rear of the engine. The lower CPR allowed the high pressure turbine to be reduced to one stage. The gas properties entering the low pressure turbine allowed that turbine to be reduced to three stages. To shorten the engine further and provide a good flow path, the combustor was reversed and placed around the turbine case.

The weight breakdown by components indicates that removal of the compressor, two turbine stages, and the general shortening of the engine would result in a 542kg (1200 pound) reduction in weight. This is a 17 percent bare weight reduction. It was assumed that the heat exchanger would fit radially within the bounds determined by the physical diameter of the low pressure turbine exit case. What remained to be determined was the actual weight and losses due to the heat exchanger and the weight of the radial compressor.

Example of Heat Exchanger Calculations

To see if any reasonable results could be obtained, a very simple and idealistic heat exchanger was envisioned. It would have radial heat pipes, be 1.2m (4 ft.) long and have a maximum radius of 0.66m (26 in.). Thus it would just fill the space saved by the changes to the cycle. If the heat exchanger could be kept within these physical limits, it would not appreciably change the outer size or drag of the nacelle. What remained to be found was the weight and

performance of the heat exchanger.

The air side was examined first. The variables were the fin spacing, t , and the frontal parameter f . From figure 2, t is seen to be the distance between the heat pipes at radius R_2 . This radius defines the position of the shell which separates the air and gas streams. The ratio of the frontal area in the cold side to the total area defined by the maximum radius of 0.66m is called f . Thus

$$f = A_1 / (A_1 + A_2) \quad (1)$$

Initially the heat pipes were assumed to act as perfect fins without any internal losses and to be so thin that they didn't take up any of the frontal area needed for the flow. If reasonable solutions were found with these assumptions, then the problem could be investigated further.

Table II lists the total temperatures, pressures and some relationships for Mach number and velocity for the selected cycle at cruise. The selected cycle had an OPR of 10 and a T_4 of 1810K (3260R). All of the important design point parameters for this cycle are listed in table II. The velocity relations were obtained from the conservation of mass equation.

$$m = f \cdot A \cdot V = P \cdot A \cdot V / (R_a \cdot T) \quad (2)$$

Solving for the velocity V ,

$$V = m \cdot R_a \cdot T / (P \cdot A) \quad (3)$$

Total conditions are used since the Mach numbers are low. The area for the cold flow (air) is,

$$A_1 = f \cdot \pi \cdot (R_1)^2 \quad (4)$$

Taking the three cases as shown in table III, the heat that must be transferred is,

$$Q_1 = m_{3x} \cdot c_p \cdot \Delta T \quad (5)$$

where ΔT is the change in temperature of the air from station 3 to 3x.

Air side heat transfer for case 1: Assume that the temperature of the heat pipe is half way between the temperature of the air and gas at each end of the exchanger. Assume also that the temperature of the air film around the heat pipe is half way between the temperature of the heat pipe and the air. Then the film temperature (T_f) is,

$$T_{f3} = T_{55x} - T_{3x}/4 + T_{3x} \quad T_{f3x} = T_{55} - T_{3x}/4 + T_{3x} \quad (6)$$

The density (ρ) of the film is calculated from the gas law using total conditions since the Mach numbers are low.

$$\rho_3 = P_3 / (R_a \cdot T_f 3) \quad \rho_{3x} = P_{3x} / (R_a \cdot T_f 3x) \quad (7)$$

The Prandtl numbers can be found to be,

$$Pr_3 = 0.688 \quad Pr_{3x} = 0.709$$

The Reynolds number, using t as the characteristic dimension, are

$$Re_3 = \rho_3 \cdot V_3 \cdot t / \mu_3 \quad Re_{3x} = \rho_{3x} \cdot V_{3x} \cdot t / \mu_{3x} \quad (8)$$

From Chapman, reference 8, equation 8.19, the Nusselt number is,

$$Nu = h \cdot t / k = 0.023 (Re)^{0.8} (Pr)^{0.4} \quad (9)$$

This can be solved for the heat transfer coefficient, h

$$h = k (0.023) (Re)^{0.8} (Pr)^{0.4} / t \quad (10)$$

Therefore,

$$h_3 = (\text{constant}) \cdot t^{0.2} \quad h_{3x} = (\text{constant}) \cdot t^{0.2} \quad (11)$$

Using the log mean relationship as a first estimate,

$$Q_2 = Ar ((h_3 x) \Delta T_3 x - (h_3) \Delta T_3) / (\ln((h_3 x) \Delta T_3 x / (h_3) \Delta T_3)) \quad (12)$$

where Ar is the surface area required and where the ΔT 's are the difference between the temperature of the heat pipe and the air stream. Substituting the values of h just calculated and setting the results equal to the value of Q_1 from equation 5,

$$Q_2 = Ar \cdot (t)^{0.2} \cdot \text{constant} = Q_1 \quad (13)$$

Solving for area required (Ar) in case 1,

$$Ar = \text{constant} / t^{0.2} \quad (14)$$

Air side heat transfer for cases 2 and 3: For cases 2 and 3, the only difference is V_3 and V_{3x} . These values show up in the Re and therefore, in the Nu and h to the 0.8 power. Reworking the problem leads to,

$$Ar = \text{constant} / t^{0.2} \quad \text{for case 2} \quad (15)$$

$$Ar = \text{constant} / t^{0.2} \quad \text{for case 3} \quad (16)$$

For three values of t and three values of ρ , the nine required areas are shown in table IV. Assuming the

cross-sectional area of the flow to be square instead of circular allows the available surface area (A_a) of the fins to be calculated more easily. The square must have sides of length L_x where,

$$L_x = (\pi(R_1)^2)^{\frac{1}{2}} \quad (17)$$

since the maximum circular area was also $\pi(R_1)^2$. Each fin has a height of $(\pi(R_1)^2)^{\frac{1}{2}}$ and a length of 1.2m. The number of fins is $(\pi(R_1)^2)^{\frac{1}{2}}/t$ and there are two sides to each fin. Therefore,

$$A_a = 2 \cdot f \cdot (\text{height}) \cdot (\text{length}) \cdot (\text{number of fins}) \quad (18)$$

or

$$A_a = 2 \cdot f \cdot L_x \cdot \pi(R_1)^2 / t \quad (19)$$

For the values of t and f selected in this study, table V shows the values of A_a , the available area.

RESULTS AND DISCUSSION

Cycle Results

The results of the cycle analysis are plotted in figure 3. It is rather obvious from part (a) of the figure that only small gains in SPC can be expected at an ϵ of 0.70. Increasing OPR beyond 12 doesn't make much sense because the additional gains are small. (It was desired to restrict the CPR to 5 if possible so a one stage centrifugal compressor could be used.) For reasons of heat exchanger size, the high levels of T_4 were best. Thus, the engine selected for the heat exchanger design was one with a OPR of 10 and a T_4 of 1810K (3260R). If the design objectives of $\epsilon=0.70$ and total pressure loss through the exchanger of 6 percent could be achieved, this engine would have an uninstalled cruise thrust of 40600N (9142pounds) and an SPC of 0.0781kg/hr/N (0.771 hr⁻¹). This would be an improvement of 4 percent in thrust and 3.3 percent in SPC compared to the reference turbofan engine.

Normally this would not create much excitement because in a parametric study, the inputs are often not known any more accurately than 3 percent. However, working with a specific study engine improves the accuracy of the delta's in weight and drag. Also, the extra thrust might allow the engine to be scaled down in size and weight thus saving more fuel than the 3.3 percent in SPC might at first indicate.

The takeoff data shown for this engine selection in part (b) of figure 3, reveals a 1 percent improvement in thrust and a 3 percent improvement in SFC. Since it was decided that the heat exchanger would be sized at cruise, takeoff is an off-design point. Because off-design heat exchanger estimates were not warranted at this time, part (b) is only an estimate. In real life the regenerator probably would not perform as well at takeoff as at cruise. This would lead to higher SFC's and possibly higher levels of thrust if the pressure drops did not get too large. It was decided that this could be calculated at a later time if desirable.

Geometry of the Heat Exchanger

The results shown in tables IV and V are plotted in figure 4. A quick look at this figure will show that no common solution exists between area required and area available over the range of ϵ and t examined. Even when the entire area ($\epsilon=1.0$) is used to pass just the airflow, A_a and A_r are an order of magnitude apart if $t=0.0164m$ (1/4 in). Larger values of t make the difference between A_a and A_r even more pronounced. Thus, small values of t seem desirable. However, if the actual width of the heat pipes is taken into account, this could easily block the entire flow area. Of course, only the air side heat transfer has been considered. The gas side will cause even larger problems for packaging since the density is lower. The heat pipe experts at Lewis Research Center suggested that the heat pipes should be made of stainless steel with 0.076cm (0.030 in) thick walls for long life. If the entire area was filled with heat vanes 0.64cm wide (1/4 in) made of 0.0118cm thick stainless steel, the weight would be 3170kg (7000 lbs.). This is more than the weight saved by rearranging and redesigning the engine. To this weight, of course, must be added the weight of the ducting and the single stage high compressor. Thus, within the size constraints assumed in this study, the additional weight of the heat exchanger alone would more than offset the meager SFC gains. This is easily demonstrated for any reasonable sensitivity of fuel or takeoff gross weight (TOGW) to changes in SFC and weight. This is not to say that A_a could not have been made larger. In fact, by just extending the length of the heat exchanger, A_a could be made equal to A_r . However, this would just make the weight picture look even worse. Using Titanium would reduce the weight by almost a factor of two, but this would still not be enough of a weight reduction.

The pressure drop is a function of velocity squared.

The velocity is a function of the flow area available to pass the required mass of air or gas. Since the weight and size of the regenerator are so obviously out of step with the desired values, a detailed design was not carried out in this study. However, if the required flow area could be achieved so the velocity could be kept low, there is no reason that the pressure drops desired could not be achieved. On the air side, the velocity would have to be between 45 and 61 m/sec (150 to 200 ft/sec). On the gas side, the velocity could be somewhat greater since the density is lower.

It seems clear from this analysis that to make this type of heat exchanger work on this engine would require a very large package. This seems to be verified by a closer examination of reference 4. In reference 4 the heat pipes used were much advanced in weight and construction features compared to those the Lewis experts would expect to see put to use in a real application such as this one. Yet in reference 4, the conclusion was that the heat exchanger would weigh more than a conventional heat exchanger. It was also found in reference 4 that a heat exchanger frontal area of 0.55m^2 (6 ft 2) was needed to pass 2.2kg (5 lbs) of flow with reasonable losses and performance. In contrast, the engine studied here tried to force 33 kg (74 lbs) of flow through 1.37m^2 (14.75 ft 2). It just can not be done reasonably. This engine would probably need about 9.29m (100 ft) of frontal area in the heat exchanger to pass the flow properly. This would require a radius of 1.8m (5.6 ft). This means the engine diameter in the area of the turbine exhaust would be 2.5 times larger than it is now even before the fan duct flow requirements are even considered. The weight and drag penalties of such a system would more than offset the meager SPC gains shown in this study. Also, heat pipes of this length are impractical. Therefore, the heat pipes would have to be oriented in a different way or the flow could be split so two or more heat exchangers were operating in parallel.

CONCLUDING REMARKS

A preliminary evaluation was made of a regenerative turbofan engine using heat pipes for the heat exchanger. The heat pipes used sodium for the working fluid and the effectiveness of the heat exchanger was fixed at 0.70, a typical value estimated by C. Silverstein in a detailed analysis of advanced heat pipes for this type of application. Turbine-rotor-inlet temperature was varied from 1480 to 1910 K and overall pressure ratio was varied from 6 to 12. The heat exchanger pressure loss was assumed

to be 3 percent on the cold side and another 3 percent on the hot side. A total of 16 engines were compared against a reference turbofan engine with an overall pressure ratio of 25 and a turbine rotor-inlet temperature of 1480 K. The bypass ratio of all the engines was held at 4.5 and the fan pressure ratio was fixed at 2.0, as in the reference engine.

Of the 16 regenerative cycles considered, one was selected for a design point study. This cycle had a turbine-rotor-inlet temperature of 1810 K and an overall pressure ratio of 10. It had a 3.3 percent better specific fuel consumption than the reference turbofan (uninstalled) at a cruise condition of Mach 0.98 and 11.6 km.

In the actual calculations for the heat exchanger, the assumptions were highly idealized in order to determine if any fuel savings were possible when the size and weight of the heat exchanger was included. The heat pipes were assumed to be stainless steel fins with a wall thickness of 0.076 cm (0.03 in). The heat transfer calculations were done on the air side first to see if that part of the heat exchanger could be made to fit behind the low-pressure turbine.

From the air side calculations it was found that not enough frontal area was available to pass the airflow desired at the desired flow conditions. The fin surface area required was much greater than the available fin surface area. This situation could have been resolved by making the heat exchanger much longer, but the weight of the heat exchanger was already excessive. The gas side heat exchanger calculations were not completed as a result of this finding.

The most significant input to this study would seem to be the heat pipe weight. The pipes were assumed to be made of stainless steel with walls 0.076cm thick. This input is a direct result of the Lewis Research Centers heat pipe experts. They felt that this type and thickness of material was needed to insure long, trouble free life. Authors such as C. C. Silverstein in ref. 1, suggest the use of heat pipes with wall thicknesses of from 0.0076 to 0.0152cm. Such wall thicknesses would reduce the weight of the heat pipes by a factor of 5 to 10. Combining this with the use of Titanium would reduce the weight even further. It would seem that these types of breakthroughs will be necessary if heat pipes are to come close to competing with other types of heat exchangers. Other advances proposed in ref. 1 include the use of two-zone capillary wicks in the pipes and working fluids such as cesium and potassium.

On the negative side, ref. 1 points out that the cost of the heat pipes could be excessive. In ref. 1 the heat pipe cost ranged from \$0.70 to \$1.40 per pipe for production rates of 10 million per year. Today, the cost of much

simpler heat pipes in small quantity is more like \$ 70.00.

It is the conclusion of this study that the size, weight and cost of this type heat exchanger make it impractical for large turbofan engines at this time. If weight and size are of secondary importance, or if heat pipe weight and cost technology is significantly improved, this type heat exchanger could possibly serve very satisfactorily in some applications due to the high reliability associated with heat pipes. Applications such as ground power plants, ships, trains, and maybe even trucks, buses, and cars might prove much more acceptable than large turbofan engines. Heat pipe heat exchangers might be used in flight applications such as small helicopters where the bulge could be easily hidden. Another less probable application might be small turboprop engines. If the engine airflow is small and the cruise speed is moderate, the shortcomings of weight and drag might be overcome by the improved fuel consumption. The cost of the heat pipes is a significant problem that would need serious consideration before any attempt to use them in great quantity could be considered.

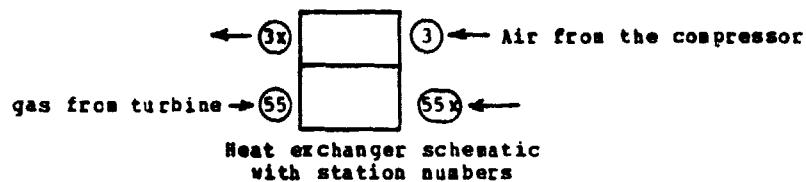
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TABLE I.- RANGE OF DESIGN PARAMETERS

Mach number,	.98				
Altitude, ka, (ft)	11.6 (38000)				→
Fan corrected airflow, kg/sec, (lb/sec)	543 (1193)				→
Fan pressure ratio	2.0				→
Fan efficiency	.852				→
Bypass ratio	4.5				→
Overall pressure ratio	25	6 to 12			→
Compressor efficiency	.862	.832			→
Combustor P/P, percent	6				→
Combustor efficiency	1.0				→
T ₄ , K, (R)	1480 (2660)	1480 (2660)	1590 (2860)	1700 (3060)	1810 (3260)
T ₄ on a 306K(550R) hot day K, (R)	1700 (3060)	1700 (3060)	1810 (3260)	1920 (3460)	2030 (3660)
High pressure turbine efficiency	.90				→
Low pressure turbine efficiency	.904				→
Nozzle P/P, percent	1.2				→
Nozzle, Cv	0.98				→
Turbine cooling flow, percent of compressor flow	6.32	4.0	7.0	10.0	13.0
Heat Exchanger:					
P/P, Air side, percent	---	3.0			→
P/P, gas side, percent	---	3.0			→
€	---	0.7			→

TABLE II.- CRUISE DESIGN POINT DATA



	Units	
	SI	English
T ₃ , K, (R)	550	(989)
P ₃ , std. atmospheres	3.73	3.73
m ₃ =m _{3x} , kg/sec, (lb/sec)	33.6	(74.3)
V ₃ , m/sec, (ft/sec)	16.22/A ₃	(573/A ₃)
Mach number 3	1.975 10 V ₃	(6.48 10 V ₃)
T _{3x} , K, (R)	1038	(1868)
P _{3x} , std. atmospheres	3.62	3.62
V _{3x} , m/sec, (ft/sec)	30.64/A _{3x}	(1082/A _{3x})
Mach number 3x,	1.30 10 V _{3x}	(4.72 10 V _{3x})
T ₅₅ , K, (R)	1248	(2243)
P ₅₅ , std. atmospheres	0.634	0.634
m ₅₅ =m _{55x} , kg/sec, (lb/sec)	39.5	(87.2)
T _{55x} , K, (R)	863	(1555)
P _{55x} , std. atmospheres	0.615	0.615
Turbine cooling flow, kg/sec, (lb/sec)	5.02	(11.1)

TABLE III.- THREE CASES

Case		A1, m ² , (ft ²)	V3, m/sec, (ft/sec)	MN3	V3x, m/sec, (ft/sec)	MN3x
1	.1	.136, (1.47)	106, (348)	0.22	200, (657)	0.31
2	.5	.685, (7.37)	20.7, (68)	0.04	39.9, (131)	0.062
3	1.0*	1.37, (14.75)	10.4, (34)	0.02	19.8, (65)	0.031

* Limit

TABLE IV.- AREA REQUIRED, A.R. m², (ft²)

t, m, (in)	t, m, (ft)	Case-1, =0.1	Case-2, =0.5	Case-3, =1.0
0.0064, (.25)	.363, (.416)	706, (7600)	2573, (27700)	4552, (49000)
0.152, (6.0)	.686, (.87)	375, (4030)	1360, (14650)	2406, (25900)
0.305, (12)	.788, (1.0)	325, (3500)	1184, (12750)	2090, (22500)

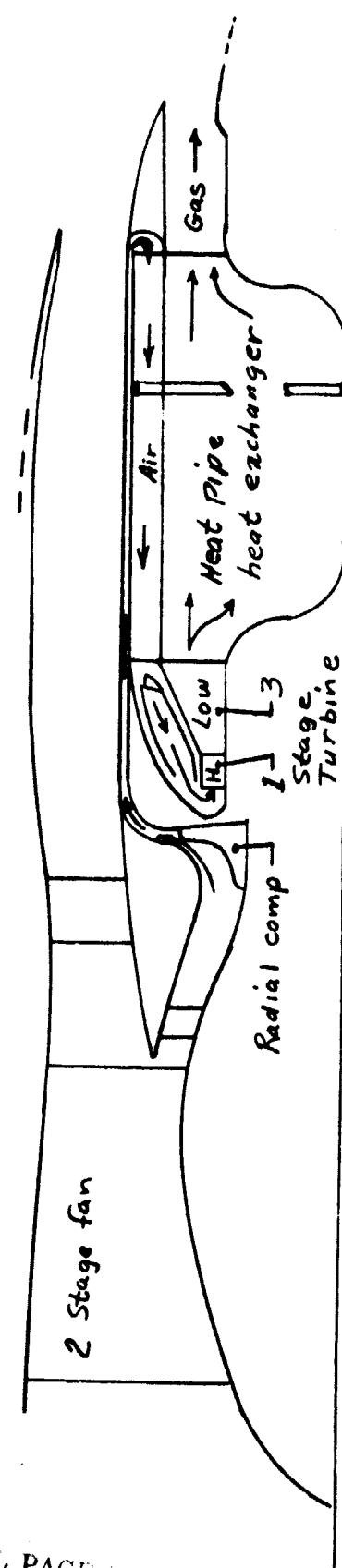
TABLE V.- AREA AVAILABLE, A.A. m², (ft²)

t, m, (in), (ft)	Case-1, =0.1	Case-2, =0.5	Case-3, =1.0
.0064, (.25), (.0203)	52.6, (566)	263, (2830)	525, (5650)
.152, (6), (0.5)	2.19, (23.6)	11.0, (118)	21.9, (236)
.3048, (12), (1)	1.09, (11.8)	5.48, (59)	11.0, (118)

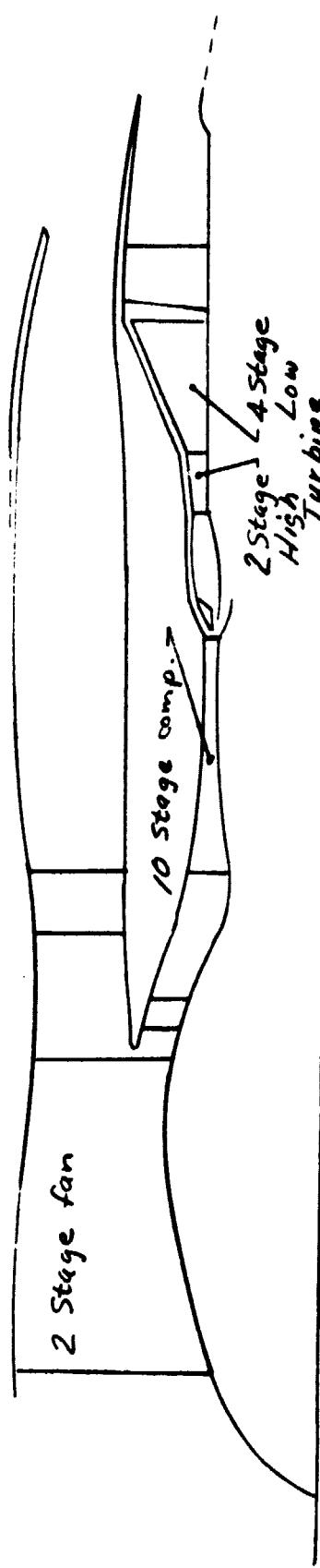
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Figure 1 - Flow path sketch of reference and regenerative turbofan engines.

a) Regenerative turbofan



b) Reference turbofan



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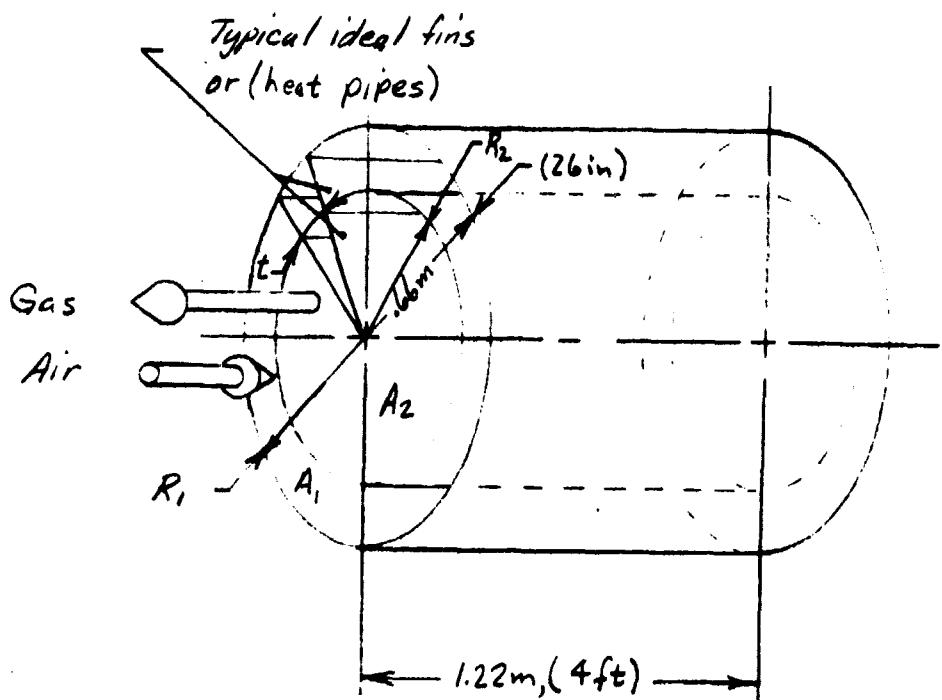


Figure 2:- Sketch of heat pipe heat exchanger

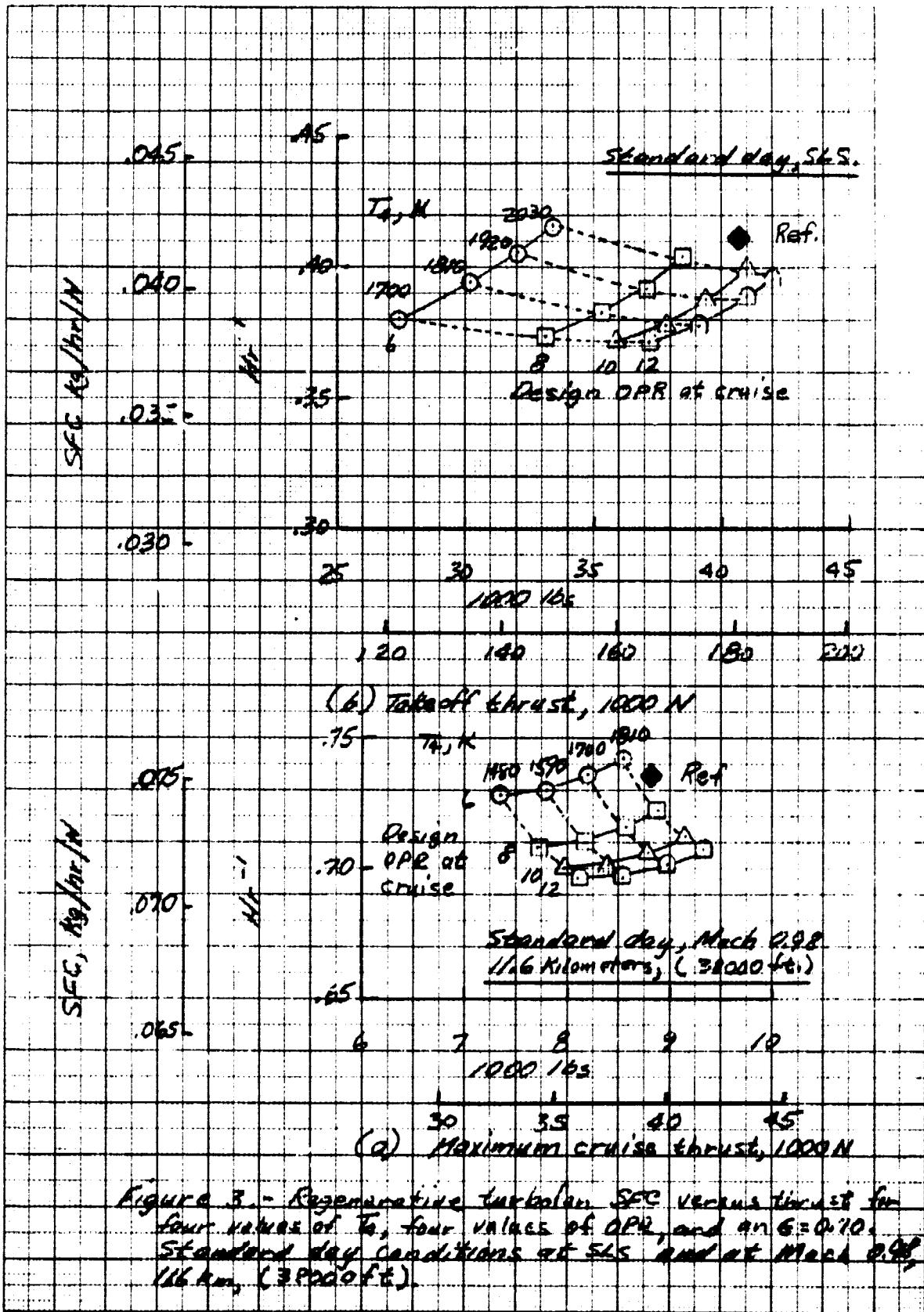


Figure 3. - Regenerative turbolow SEC versus thrust for four values of T_0 , four values of OPR, and an $\epsilon = 0.70$. Standard day conditions at SLS and at Mach 0.98 (116 Km, (38000 ft)).

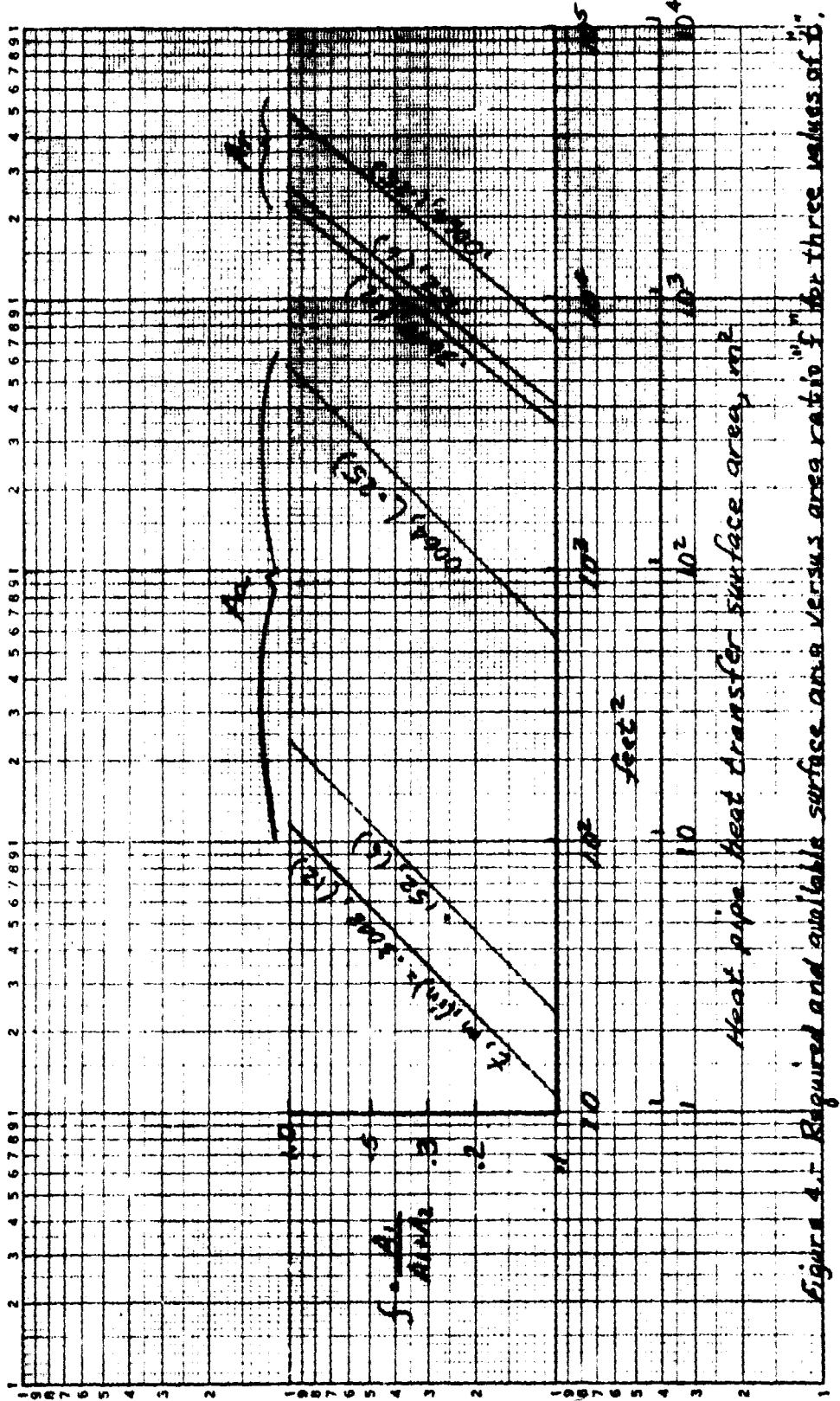


Figure 4.- Required and available surface area versus area ratio "f" for three values of "f".